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(54) Gas turbine cycle incorporating simultaneous, parallel, dual-mode heat recovery.

(57) The cycle includes water-injection, steam-injection, recuperation (or regeneration) and waste-heat boiler heat recovery in an arrangement that provides high thermal efficiency, flexible operation in a cogeneration plant and favorable capital cost in relation to thermodynamic performance when compared to currently practiced cycles. In the present cycle, the sensible enthalpy of the exhaust gases between turbine exit and stack is used to simultaneously and in-parallel heat both air and water/steam. A smaller amount of water is boiled than in the known Cheng cycle, in which the exhaust heat is used only to heat water/steam. Thus, the latent heat exhausted at the stack in the present cycle is lower than that for the Cheng cycle resulting in higher efficiency.

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EUROPEAN SEARCH REPORT

Application Number

EP 88 10 0532

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.4)
A	EP-A-0 081 996 (MITSUBISHI GAS CHEMICAL CO.) * Absract; page 8, lines 13-23; figure 4 *	1-17	F 01 K 21/04 F 02 C 3/30
A	DE-C- 717 711 (MARTINKA) * Page 1, line 38 - page 2, line 37; figure 1 *	1-17	
A	EP-A-0 207 620 (IMPERIAL CHEMICAL INDUSTRIES) * Abstract; page 13, line 17 - page 14, line 30; figure 1 *	1-17	
A	DE-C- 718 197 (MARTINKA) * Page 2, line 85 - page 3, line 18; figure *	1-17	
A	US-A-2 678 532 (B. MILLER) * Column 3, lines 8-59; figure 1 *	1-17	
A	FR-A-2 577 990 (ELECTRICITE DE FRANCE) * Abstract; figure 1 *	1	TECHNICAL FIELDS SEARCHED (Int. Cl.4)
A	EP-A-0 150 990 (FLUOR CORP.)		F 01 K F 02 C F 22 B
A	US-A-2 959 005 (ZABA)		
A	US-A-2 186 706 (MARTINKA)		

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	30-11-1988	ERNST J.L.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
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(54) Gas turbine cycle incorporating simultaneous, parallel, dual-mode heat recovery.

(57) The cycle includes water-injection, steam-injection, recuperation (or regeneration) and waste-heat boiler heat recovery in an arrangement that provides high thermal efficiency, flexible operation in a cogeneration plant and favorable capital cost in relation to thermodynamic performance when compared to currently practiced cycles. In the present cycle, the sensible enthalpy of the exhaust gases between turbine exit and stack is used to simultaneously and in-parallel heat both air and water/steam. A smaller amount of water is boiled than in the known Cheng cycle, in which the exhaust heat is used only to heat water/steam. Thus, the latent heat exhausted at the stack in the present cycle is lower than that for the Cheng cycle resulting in higher efficiency.

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brackets to designate component nomenclature as indicated on Fig. 1a. Motive fluid, typically ambient air, is compressed from state [1] to state [CD] in a compressor 1. The compressed air then enters aftercooler la where it is evaporatively cooled by water injection thereby reducing its temperature and increasing the mass flow rate of motive fluid, which exits the aftercooler at state [2B] and flows into a low-temperature recuperator (LTR) 2 where it is heated to a temperature close to the saturation temperature at the gas turbine combustor pressure, state [2C]. The motive fluid is then mixed with steam and the mixture flows through a high temperature recuperator (HTR) 3 where it is heated to state [2] before being conveyed to the combustor 3a. In most cases the combustor 3a will be a directly fired design as commonly used in gas turbine engines but could also be an indirectly fired heat exchanger. Heated motive fluid leaves the combustort at state [3] and enters a gas turbine 3b, where it is expanded to close-to-ambient pressure, state [4], thereby producing work to drive the compressor 1 and external load. The hot exhaust motive fluid as state [4] is then conveyed to the heat recovery system where it flows in counterflow heat exchange relationship against the cooler compressed motive fluid. A damper 7 (or other flow control device) is used to apportion the flow of hot exhaust gases between two parallel paths in the heat recovery system. For high work conversion efficiency, damper 7 is in the position shown by dotted lines and all the exhaust flows through HTR 3 where it is cooled down to state [5A]. The exhaust gases then pass through a boiler 6, raising steam while being simultaneously cooled down to state [5B]. They are then used to simultaneously heat in parallel the cool motive fluid through LTR 2 from state [2B] to state [2C] as well as water through a high] temperature economiser (HTE) 5 from state [wl] to state [ws], the latter being essentially at saturation temperature corresponding to the boiler presure. This simultaneous parallel heating may be accomplished in two physically separate heat exchange devices with the hot exhaust gas being apportioned between them by a flow control device, or in a single heat exchanger which includes parallel but separate flow paths for the cool motive fluid and boiler feedwater. The exhaust gases are thus cooled down to state [5] and then flow counter to the feedwater in a low temperature economiser (LTE) 4 where they are further cooled down to state [6] while heating the feedwater from state [wa] to state [wl]. They are then exhausted to the stack. Provision to partially condense the water vapor from the stack exhaust at state [6] in order to recycle the condensate may or may not be employed.

Figs. 3a through 3d present overall performance maps calculated for this cycle and compared to four types of cycle in current practice or currently undergoing development. The ordinate in these figures is gross cycle efficiency, the percentage of fuel lower heating value heat-input converted to cycle work-output. The abcissa on these plots is the gross cycle specific power per unit compressor inlet airflow. The term "gross" in the present context implies work output before accounting for mechanical friction, auxiliary power requirements or electric losses. The calculations for all cycles in these figures are based upon identical turbomachinery component assumptions. The effect of turbine cooling flows upon the cycle thermodynamics has been computed and included for all cycles based upon identical cooling technology assumptions. The set of common assumptions used is representive of current idustrial gas-turbine engines. In the calculations presented, the minimum "pinch-point" temperature difference allowed was 25°F, the minimum stack temperature allowed was 200°F, and the temperature difference between exhaust gases at [4] and motive fluid at [2] was 75°F. In the calculations whose results are presented for this cycle in Fig 3, it was assumed that all the aftercooler water was extracted through valve 9 and that valves 8 and 10 were shut. It was also assumed that the aftercooler water injection rate was such as to result in 95% relative humidity of the motive fluid at its exit, state [2B]. In the calculations presented for this cycle with evaporative intercooling (Fig. 1b) on Fig. 3b it was assumed that intercooler and aftercooler both saturate the air to 80% relative humidity at their outlet and that valves 8, 10, 14, and 16 of Fig. 1b were shut. For the results with surface intercooling shown in Fig. 3c for this cycle of Fig. 1c, the intercooler was assumed 80% effective. Priority was given to utilizing all the intercooler hot water first in the HTE 5 through the valve 16 then to the aftercooler 1a through valve 18. If this was insufficient, the makeup water to the heat engine cycle was drawn first through the valve 9 to the aftercooler 1a, then through a valve 17 to the HTE 5. Valves 8 and 10 were shut. Those arrangements were found most satisfactory for benchmark cases but the water valve settings need to be varied for any particular set of cycle parameters in order to achieve maximum efficiency, maximum specific power, or an optimum combination of efficiency, specific power and cogenerated hot water or steam.

The Cheng cycle and the present cycle both use waste heat to raise steam which augments the combustor and tubine mass flow rates without increasing the compression work. A comparison is therefore in order. Fig. 3a shows the present cycle to offer much higher thermal efficiencies, particular at lower pressure ratios where the specific power is somewhat lower than that for the Cheng cycle. At higher pressure ratios the efficiency advantage of the present cycle over Cheng's diminishes but can be regained by intercooling as shown in Figs. 3b and 3c. Figs. 4 through 8 together with the following discussion will elucidate the reasons.

Figs. 4 and 5 show the heat-recovery system temperature profile and key characteristics for two 8 MW non-intercooled systems with a pressure ratio of 9 and a turbine inlet temperature (TIT) of 2100°F. Fig. 4 is for the present system and Fig. 5 for the Cheng cycle. The temperature profiles for the exhaust gas cooling [4]-[5A]-[5B]-[5]-[6] and the composite (air/water/steam) motive fluid heating from states [wa] and [2B] to state [2] on Fig. 4 are in closer proximity than the profiles for heating water/steam only on Fig. 5. Thus, the irreversibility of the

effects. For high pressure ratio cycles with compressor intercooling, however, Figs. 3b and 3c show the present cycle to provide considerable efficiency advantages over a Cheng cycle due to the increased difference between turbine exhaust temperature and compressor discharge temperature.

Fig. 8 shows a dimensionless measure of the required heat transfer capacity per unit power output, (UA)Ta/W, where Ta denotes reference ambient temperature of 518.7 degrees Rankine and W the net cycle power output. This quantity is plotted against TIT for the cycle examples of Fig. 6. It shows that as the efficiency advantage of the present cycle over the Cheng cycle diminishes, the additional heat transfer surface required is also reduced.

It is instructive to compare the performance of this cycle with the three types of cycle whose performance is shown on Fig. 3d based upon identical turbine technology assumptions. Comparing the performance of the non-intercooled examples of this cycle shown on Fig. 3a with the recuperated cycles of Fig. 3d, one sees advantages of the order of six percentage points in efficiency and 30% in specific power. Those accrue from the greater effectiveness of this heat recovery system, which requires a little over twice the specific heat exchanger capacity (UATa/W) when compared to the recuperated cycle. The additional efficiency and specific power are likely to make the additional investment in the heat recovery system well justified. Compared to a combined cycle, the intercooled examples of this cycle shown in Figs. 3b and 3c are about 1.5 points in efficiency below the best dual-pressure-boiler combined cycles and have approximately the same specific power. They would have essentially the same efficiency as single-pressure-boiler combined cycles. The most efficient examples of this intercooled cycle shown on Figs. 3b and 3c have specific water consumption rates of about 0.5 lb/s/MW (surface intercooled) and 0.75 lb/s/MW (evaporatively intercooled) and specific heat exchanger capacities (UATa/W) of about 7 for the toal exhaust heat recovery system. Those indices are both superior to the examples of the non-intercooled cycle and are likely to justify the greater complexity in many plants. The savings in cost over a combined cycle, which has additional rotating machinery and heat exchangers is likely to favour my cycle, particularly in smaller cogeneration applications.

The present cycle allows flexible and efficient integration into a cogeneration plant as seen from the following operating regimes:

1. For maximum work conversion efficiency and no steam load, damper 7 is set in the dotted position shown in Figs. 1a, 1b and 1c resulting in the performance characteristics discussed above.

2. For light steam loads, up to the steam flow rate raised at the maximum eficiency condition 1 above, damper 7 remains in the dotted position of Fig. 1a and the steam bleed valve is opened while valve 11 is closed. The recuperative effect on preheating the air prior to the combustor maintains a high work conversion efficiency.

3. For maximum work output, albeit at reduced efficiency, and with no steam load, damper 7 is set to completely by-pass the HTR, thereby increasing the steam flow rate which may be injected into the gas turbine for increased power.

4. For maximum steam loads, albeit at reduced work output and work conversion efficiency, damper 7 is set to completely by-pass the HTR, thereby increasing the steam flow rate, all of which is bled to process and valve 11 is shut.

5. At any steam load between zero and that of Mode 4 above, one has a flexible tradeoff between work output and work conversion efficiency which may be controlled by the position of damper 7. By-passing more exhaust gas around the HTR results in increased steam injection but decreased recuperation, giving more power but less efficiency, and vice versa.

The present system thus allows additional degrees of freedom for variable load cogeneration applications over most existing gas turbine cogeneration plants. In cogeneration applications where superheated steam is require, a separate superheater 3a may be used in-parallel with the HTR such as shown in the variation of Fig. 9.

Other variations and arrangements of gas turbine cycles embodying my principle of simultaneous, parallel, dual-mode heat recovery will become obvious to those skilled in the art, and it is intended to claim all those which fall within the scope of this invention.

Claims

1. Dual mode heat recovery system disposed in heat exchange relation with exhaust gases from a gas turbine plant comprising:
a boiler fed from a feedwater supply means, the boiler and feedwater supply means both in heat exchange relation with the exhaust gases in a boiler/feedwater supply path;
recuperator means for conveying motive fluid in counterflow heat exchange relation with the exhaust gases in a recuperator path;
means for mixing boiler output with the motive fluid before combustion;
the boiler/feedwater supply means and the recuperator means comprising parallel and separate flow paths within the exhaust gas flow; and
flow control means for apportioning the exhaust gas flow between the boiler/feedwater supply path and the recuperator path.

2. The heat recovery system of claim 1 further including an evaporative aftercooler for cooling and increasing the mass flow rate of the motive fluid before entering the recuperator means.

3. The heat recovery system of claim 1 or claim 2 wherein the recuperator means includes a first, low temperature recuperator disposed

with the compressed motive fluid to cool it by evaporation and augment its mass flow rate;
means to convey the resulting motive fluid to recuperator means where the motive fluid is heated to close to water saturation temperature at its pressure by causing it to flow in counterflowing heat-exchange relationship with exhaust gases from the gas-turbine; a portion of the energy of said exhaust gases between the same temperature levels being used simultaneously and in parallel to heat liquid water flowing in counterflow heat-exchange relationship within a first economiser means to heat liquid feedwater to a temperature below that of the motive fluid leaving aftercooler means by causing it to flow in counterflowing heat exchange relationship with the turbine exhaust gases after said exhaust gases have flowed through both recuperator means and first economiser means;
second economiser means to heat liquid water discharged from the first economiser means to a temperature close to saturation temperature of water at the pressure of the compressed motive fluid by causing it to flow in counterflowing heat exchange relationship with turbine exhaust gases between the same temperature levels as those flowing in heat exchange relationship with the motive fluid in the recuperator means;
boiler means to boil liquid water which has already been heated in the first and second economiser means by causing it to flow in heat exchange relationship with the turbine exhaust gases, before said gases have been used to effect simultaneous heat transfer in recuperator means and economiser means; means to convey and introduce steam raised in boiler means into superheater means in which steam is superheated by causing it to flow in counterflowing heat exchange relationship with a portion of the turbine exhaust gases before said gases have been used to effect heat transfer to boiler means, this portion being controlled by flow-control means, the remaining portion of said gases being used to effect heat transfer to motive fluid in recuperator means, said heat transfer being effected simultaneously and in parallel between the same approximate temperature levels of said gases;
recuperator means for heating the motive fluid by passing it in counterflow heat exchange relationship with the portion of turbine exhaust gases not used to effect heat transfer to superheater means and before said gases have been used to effect heat transfer to boiler means;
means to convey the heated motive fluid leaving recuperator means as well as steam leaving superheater means to combustor means to raise the temperature of the mixed motive fluid; means to convey said motive fluid to turbine means to expand siad motive fluid and extract work therefrom; means to convey the exhausted motive fluid to flow control means to adjustably apportion the motive fluid discharged from the turbine means into two streams; means to convey one stream to superheater means; means to convey the other stream to recuperator means.

10. A gas-turbine power-plant cycle comprising:
compressor means to compress motive fluid, typically ambient air;
economiser means to heat liquid feedwater to a temperature close to saturation temperature of water at the pressure of the compressed motive fluid by causing it to flow in counterflowing heat exchange relationship with the turbine exhaust gases;
boiler means to boil liquid water which has already been heated in the economiser means by causing it to flow in heat exchange relationship with exhaust gases from the turbine, before said gases have been used to effect heat transfer in the economiser means;
means to convey and introduce a variable portion of the steam raised in the boiler means into the compressed motive fluid leaving the compresor means; means to convey the resulting mixture of motive fluid to recuperator means for heating the motive fluid mixture by passing it in counterflow heat exchange relationship with a portion of the turbine exhaust gases, before said portion has been used to effect heat transfer to the boiler means;
means to convey the heated mixture of motive fluid to combustor means to raise the temperature of said motive fluid; menas to convey said motive fluid to turbine means to expand said motive fluid and extract work therefrom; means to convey the exhausted motive fluid to flow control means to adjustably apportion the exhaust motive fluid discharged from the turbine means into two streams; means to convey one stream directly to boiler means; means to convey the other stream to recuperator means.

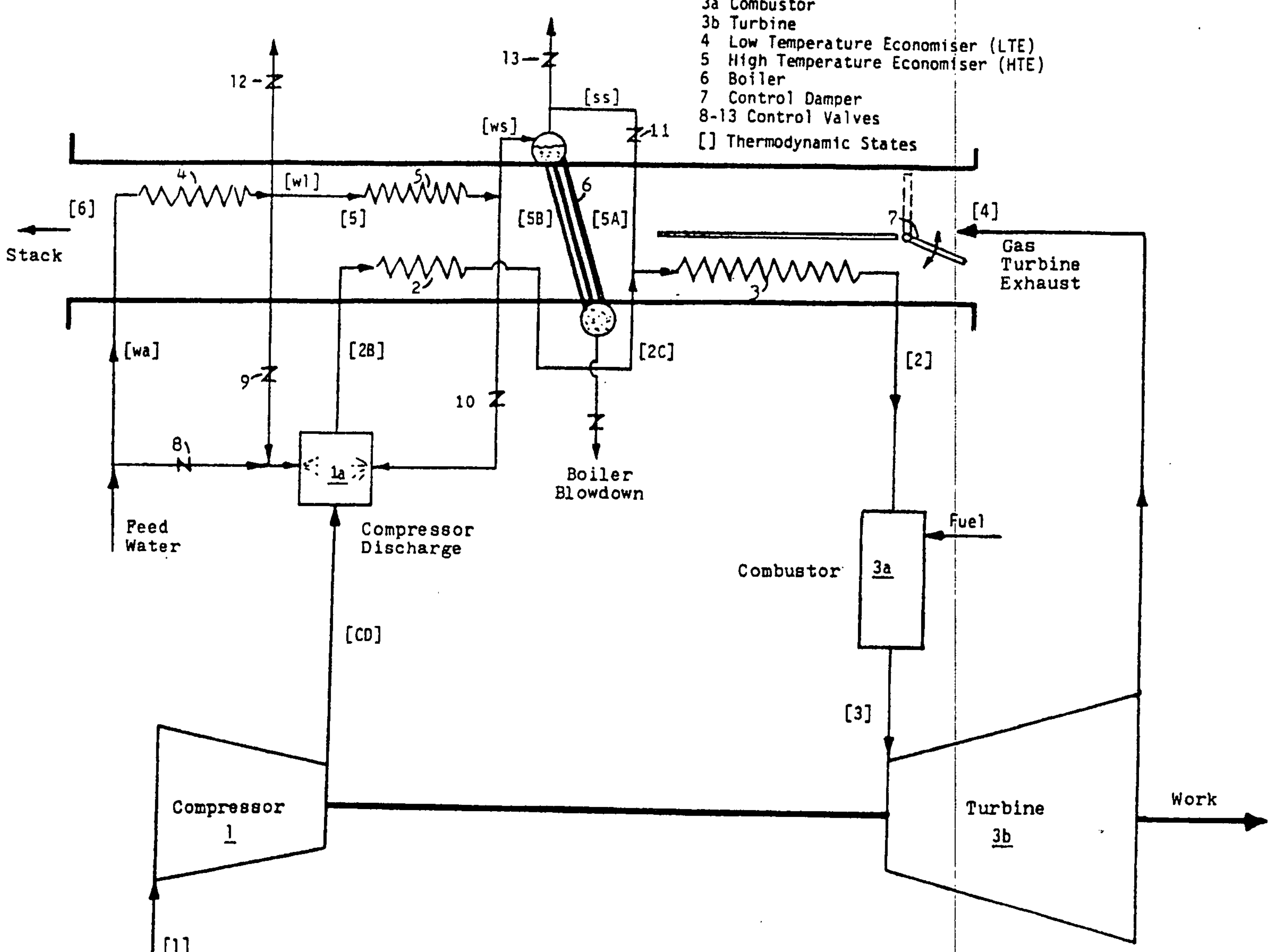
11. A gas-turbine power-plant cycle comprising:
compressor means to compress motive fluid, typically ambient air:
means to convey the compressed motive fluid to recuperator means for heating the motive fluid by passing it in counterflow heat exchange relationship with a portion of exhaust gases from the turbine, the remaining portion of the turbine exhaust gases being utilized in parallel to effect heat transfer in a heat recovery steam generator means comprising economiser means to heat liquid feedwater to a temperature close to saturation temperature of water at the pressure of the compressed motive fluid, boiler means to boil liquid water heated in the economiser means and superheater means to superheat steam raised in the boiler means; the heat source for said heat recovery steam generator being the turbine exhaust gases below the temperature of said gases exiting the recuperator means and the portion of said gases not used to effect parallel heat transfer to

the economiser by causing the water to flow in heat exchange relationship with the exhaust gases before said gases have transferred heat to the economiser means;
first superheater means adapted to receive steam from the boiler and to superheat said steam by conveying it in counterflow heat exchange relationship with the turbine exhaust gases before said gases have been utilized to effect heat transfer to the boiler means;
second superheater means adapted to receive steam from the first superheater and to further superheat said steam by conveying it in counterflow heat exchange relationship with a portion of the turbine exhaust gases before said portion of gases has been utilized to effect heat transfer to the first superheater;
recuperator means adapted to receive compressed motive fluid from the compressor and to heat said fluid by conveying it in counterflow heat exchange relationship with the portion of the turbine exhaust gases not being used to effect heat transfer in the second superheater means, and before said gases have been utilized to effect heat transfer to the first superheater means;
combustor means for receiving the compressed motive fluid from the recuperator as well as steam from the second superheater and for heating the resulting composite motive fluid;
turbine means for receiving the motive fluid from the combustor to expand said motive fluid and extract work therefrom; and
flow control means for receiving the exhausted gas motive fluid discharged from the turbine and to adjustably apportion said turbine exhaust gases into two parallel streams; means to convey one stream to the second superheater means; means to convey the other stream to thje recuperator means.

16. Gas-turbine power-plant cycle comprising:
compressor means to compress gaseous motive fluid, typically air;
economiser means adapted to heat liquid feedwater to a temperature close to its saturation temperature by conveying said feedwater in counterflow heat exchange relationship with the turbine exhaust gases;
boiler means adapted to boil liquid water from the economiser by causing the water to flow in heat exchange relationship with the exhaust gases before said gases have transferred heat to the economiser means;
superheater means adapted to receive steam from the boiler and to superheat said steam by conveying it in counterflow heat exchange relationship with a portion of the turbine exhaust gases before said gases have been utilized to effect heat transfer to the boiler means;
recuperator means adapted to receive compressed motive fluid from the compressor and to heat said fluid by conveying it in counterflow heat exchange relationship with a portion of the turbine exhaust gases, said portion of gases not being used to effect heat transfer to the superheater means and being subsequently utilized to effect heat transfer to the boiler means;
combustor means for receiving the compressed motive fluid from the recuperator as well as steam from the superheater and for heating the resultiing composite motive fluid;
turbine means for receiving the motive fluid from the combustor to expand said motive fluid and extract work therefrom; and
flow control means for receiving the exhausted gas motive fluid discharged from the turbine and to adjustably apportion said turbine exhaust gases into three parallel streams; means to convey the first stream to the superheater, means to convey the second stream to the recuperator and means to convey the third steram directly to the boiler.

17. Gas-turbine power-plant cydle comprising;
compressor means to compress gaseous motive fluid, typically air;
economiser means adapted to heat liquid feedwater to a temperature close to its saturation temperature by conveying said feedwater in counterflow heat exchange relationship with the turbine exhaust gases;
boiler means adapted to boil liquid water from the economiser by causing the water to flow in heat exchange relationship with the exhaust gases before said gases have transferred heat to the economiser means;
first superheater means adapted to receive steam from the boiler and to superheat said steam by conveying it in counterflow heat exchange relationship with the turbine exhaust gases before said gases have been utilized to effect heat transfer to the boiler means;
second superheater means adapted to receive steam from the first superheater and to further superheat said steam by conveying it in counterflow heat exchange relationship with a portion of the turbine exhaust gases before said portion of gases has been utilized to effect heat transfer to the first superheater;
recuperator means adapted to receive compressed motive fluid from the compressor and to heat said fluid by conveying it in counterflow heat exchange relationship with a portion of the turbine exhaust gases, said portion of gases not being used to effect heat transfer in the second superheater means and being subsequently utilized to effect heat transfer to the first superheater means;
combustor means for receiving the compressed motive fluid from the recuperator as well as steam from the second superheater and for heating and resulting composite motive fluid;
turbine means for receiving the motive fluid from the combustor to expand said motive fluid and extract work therefrom; and
flow control means for receiving the exhausted gas motive fluid discharged from the turbine and to adjustably apportion said turbine ex-

FIG. 1a



1 Compressor
1a Evaporative Aftercooler
2 Low Temperature Recuperator (LTR)
3 High Temperature Recuperator (HTR)
3a Combustor
3b Turbine
4 Low Temperature Economiser (LTE)
5 High Temperature Economiser (HTE)
6 Boiler
7 Control Damper
8-13 Control Valves
[] Thermodynamic States

FIG. 1C Embodiment with Surface Intercooler

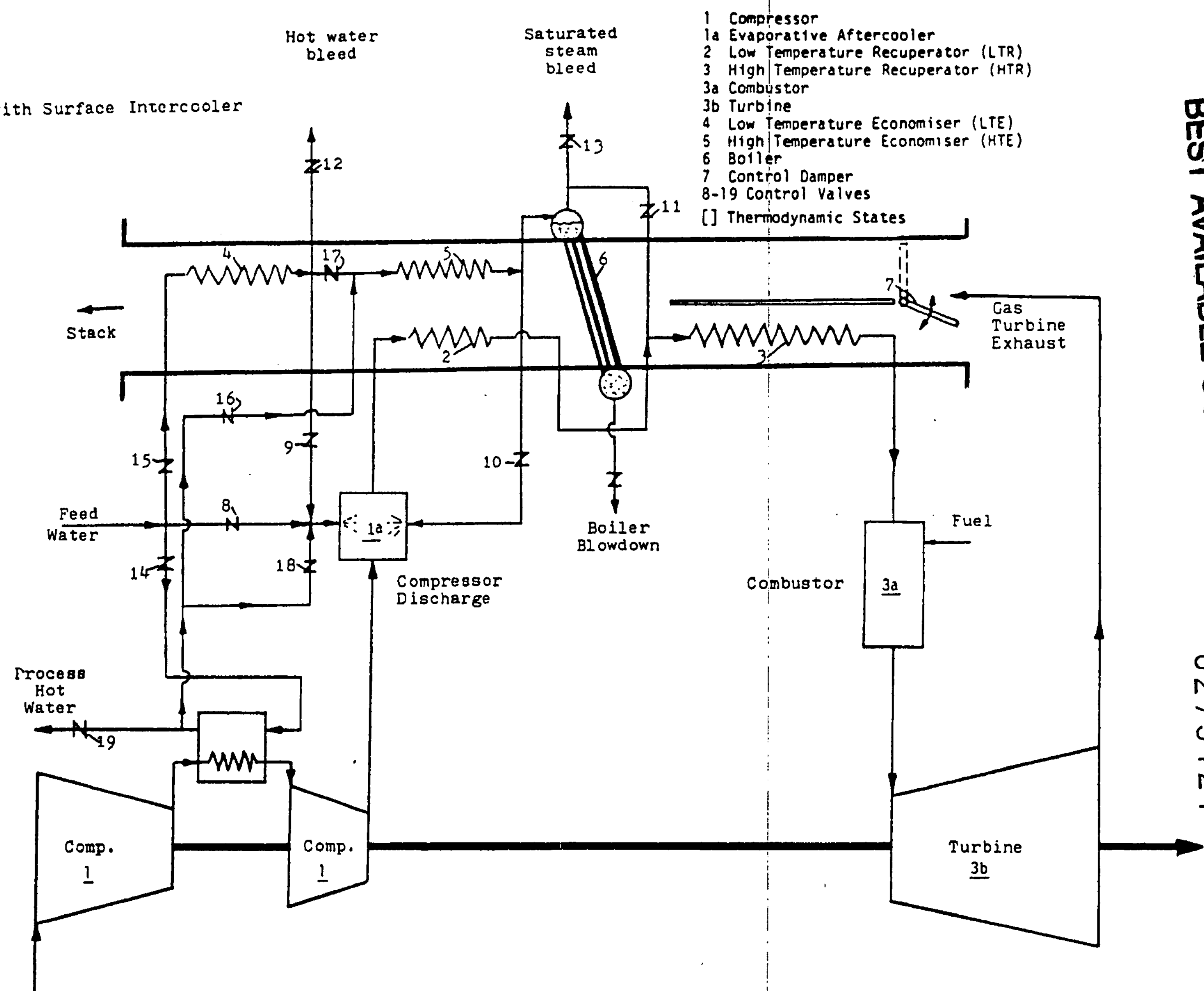
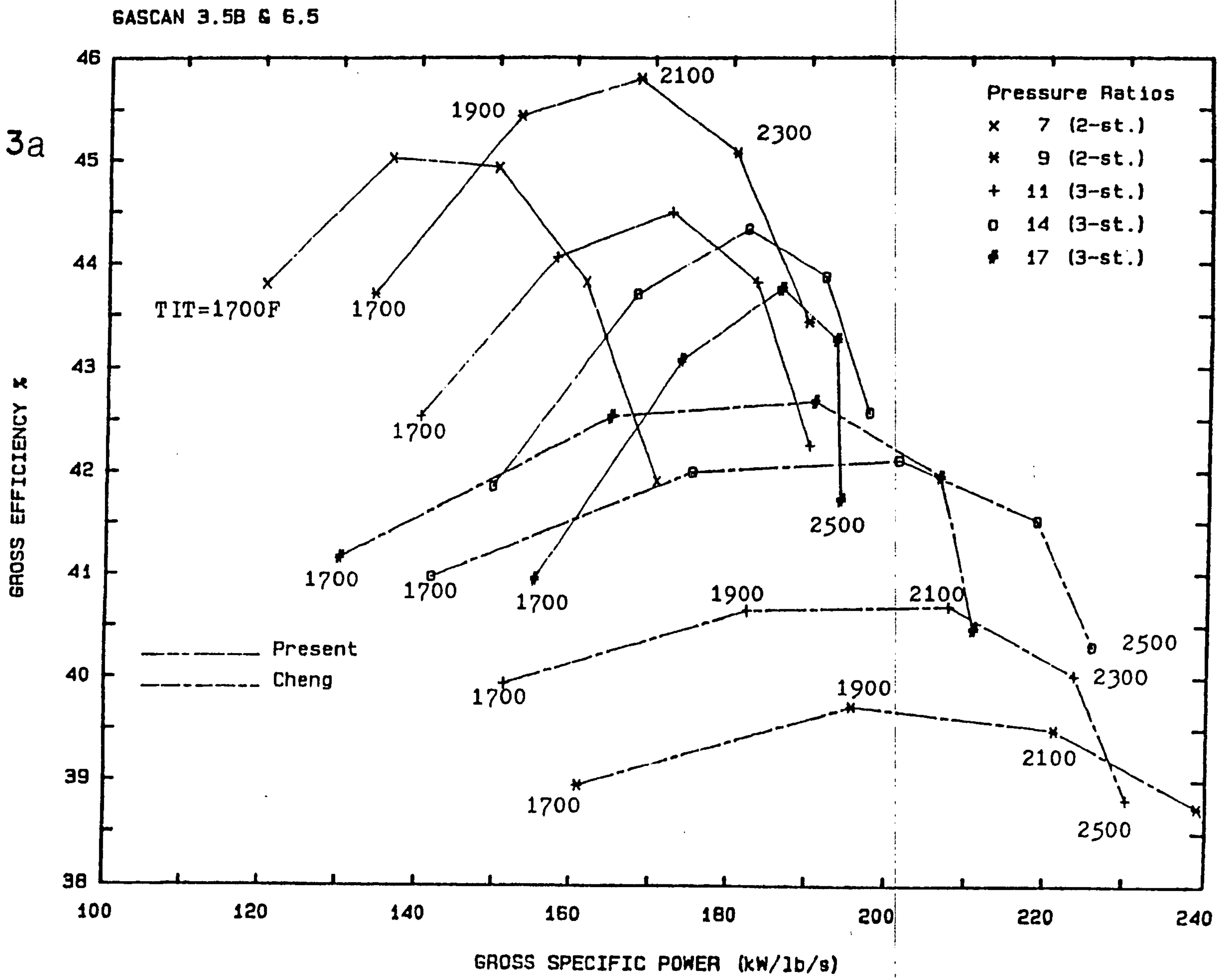


FIG. 3a


GASCAN 3.5B & 6.5
Pressure Ratios
× 7 (2-st.)
✱ 9 (2-st.)
+ 11 (3-st.)
o 14 (3-st.)
# 17 (3-st.)
TIT=1700F
1700
1900
2100
2300
2500
Present
Cheng
GROSS EFFICIENCY %
GROSS SPECIFIC POWER (kW/lb/s)
46
45
44
43
42
41
40
39
38
100
120
140
160
180
200
220
240

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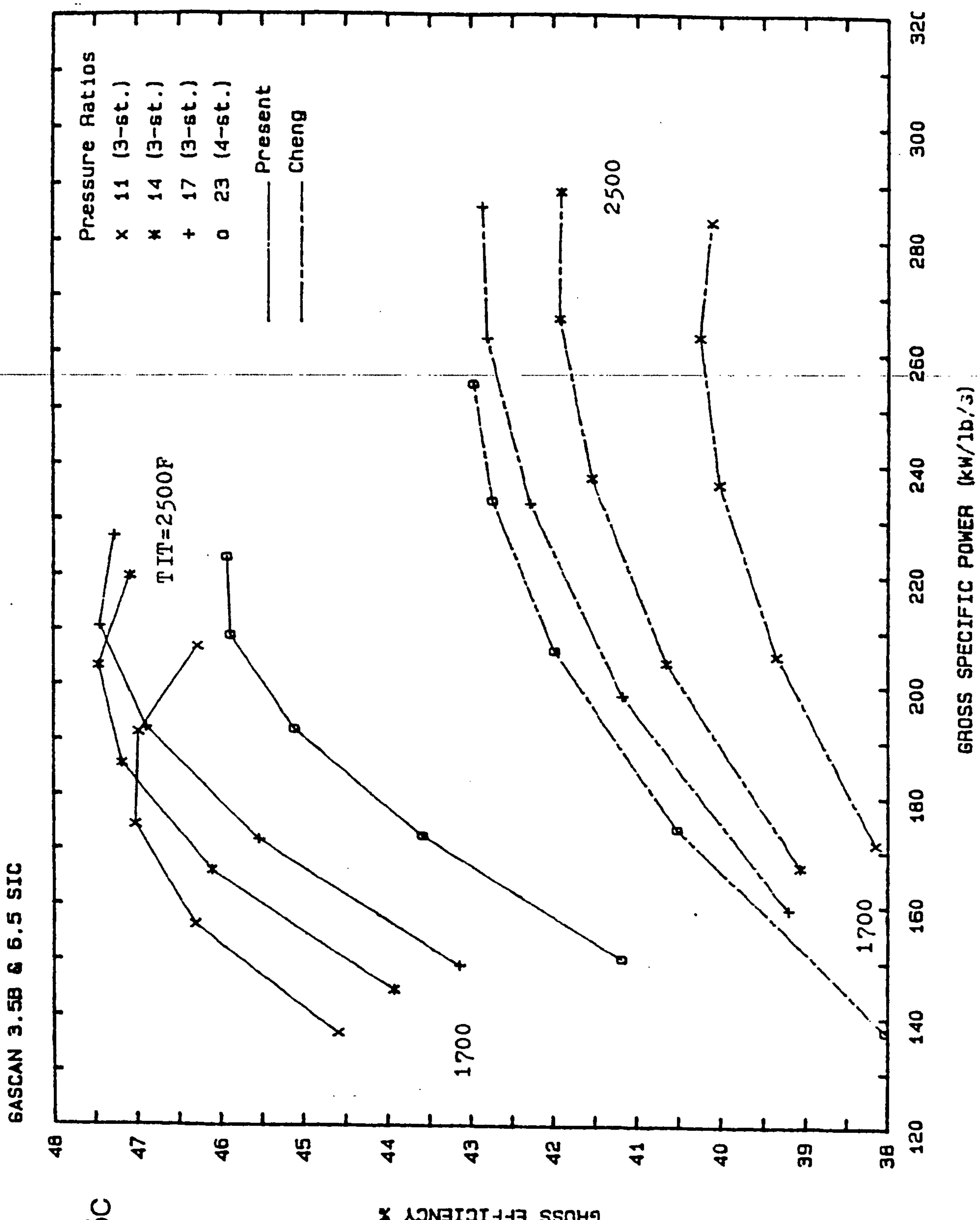

FIG. 3C

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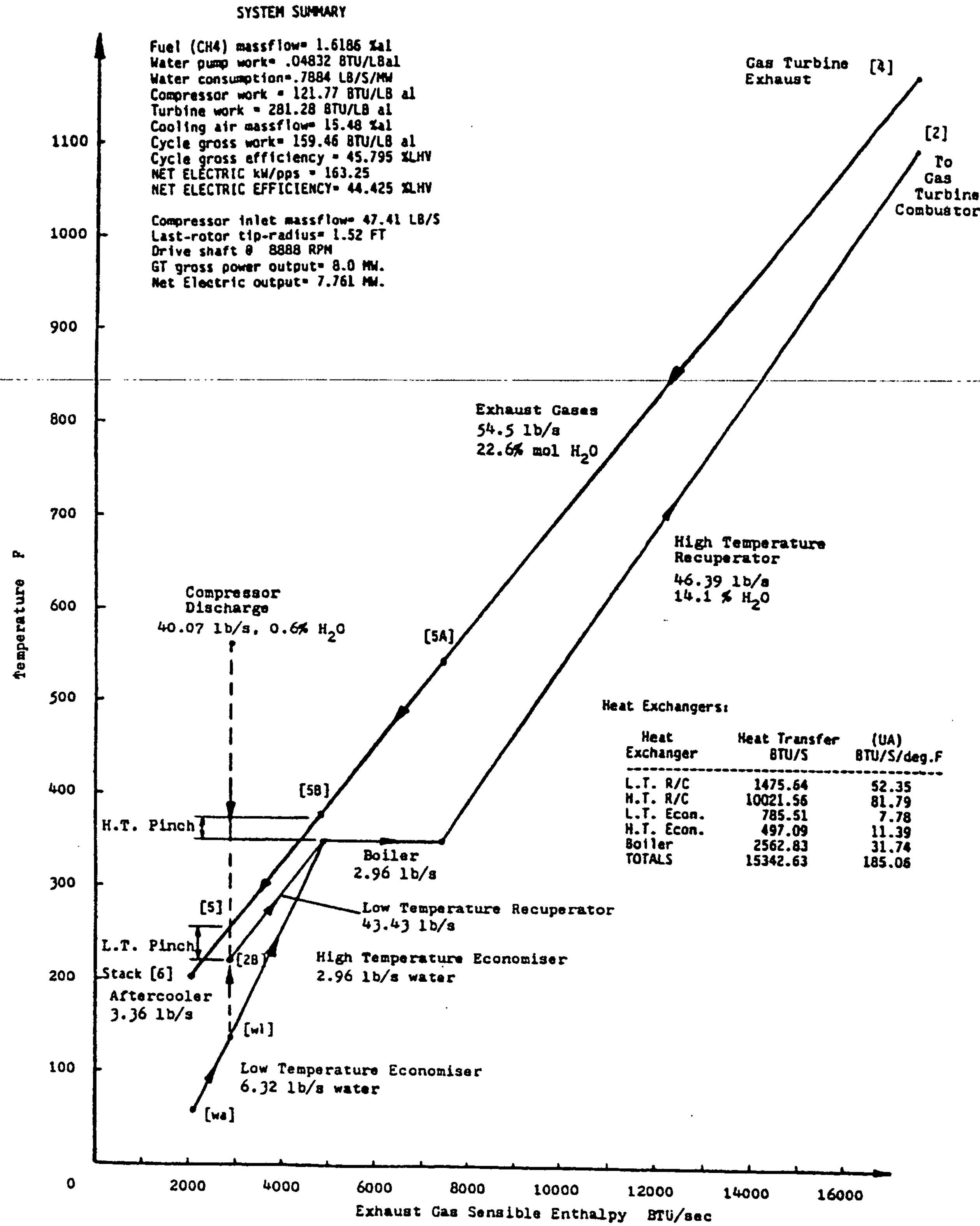

Heat Recovery Diagram for an Example of the Proposed Cycle Applied to an 8 MW Gas Turbine with a pressure ratio of 9 and combustor discharge temperature of 2100 F.

FIG. 4

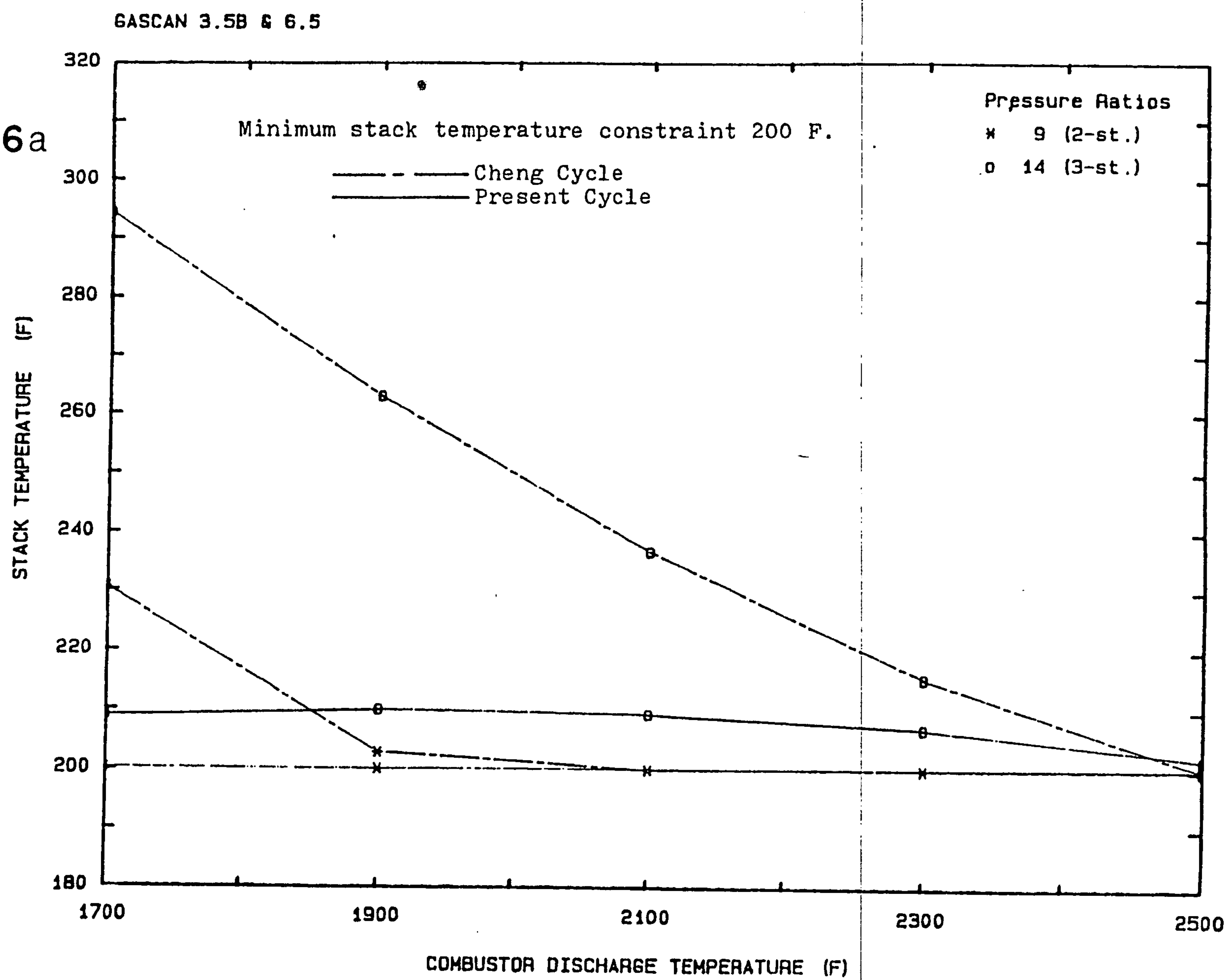
GASCAN 3.5B & 6.5
Minimum stack temperature constraint 200 F.
Cheng Cycle
Present Cycle
Pressure Ratios
* 9 (2-st.)
o 14 (3-st.)
STACK TEMPERATURE (F)
320
300
280
260
240
220
200
180
1700
1900
2100
2300
2500
COMBUSTOR DISCHARGE TEMPERATURE (F)

FIG. 6a

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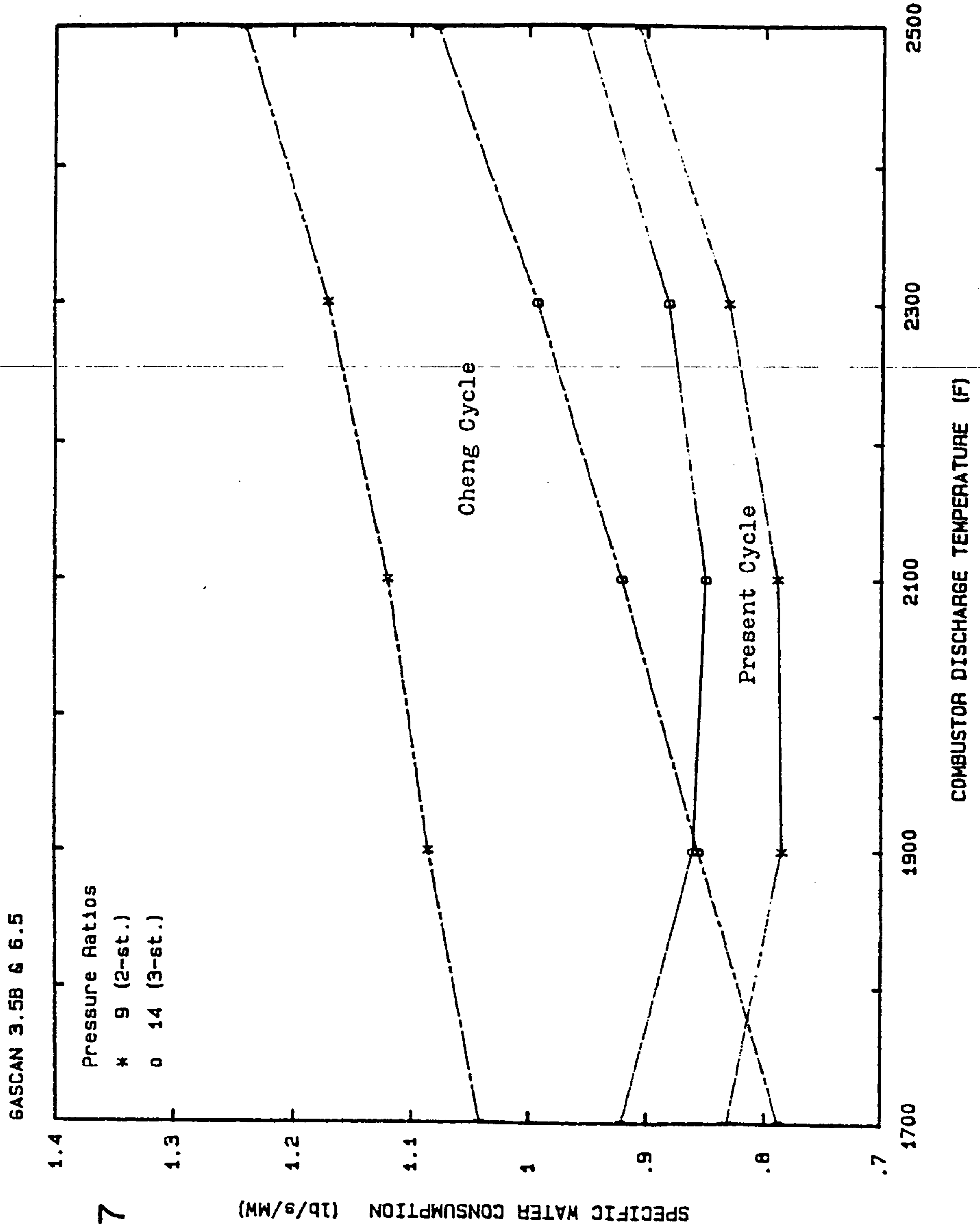

FIG. 7

1a Evaporative Aftercooler
2 Low Temperature Recuperator (LTR)
3 High Temperature Recuperator (HTR)
4 Low Temperature Economiser (LTE)
5 High Temperature Economiser (HTE)
6 Boiler
6a Superheater
7 Control Damper
8-14 Control Valves

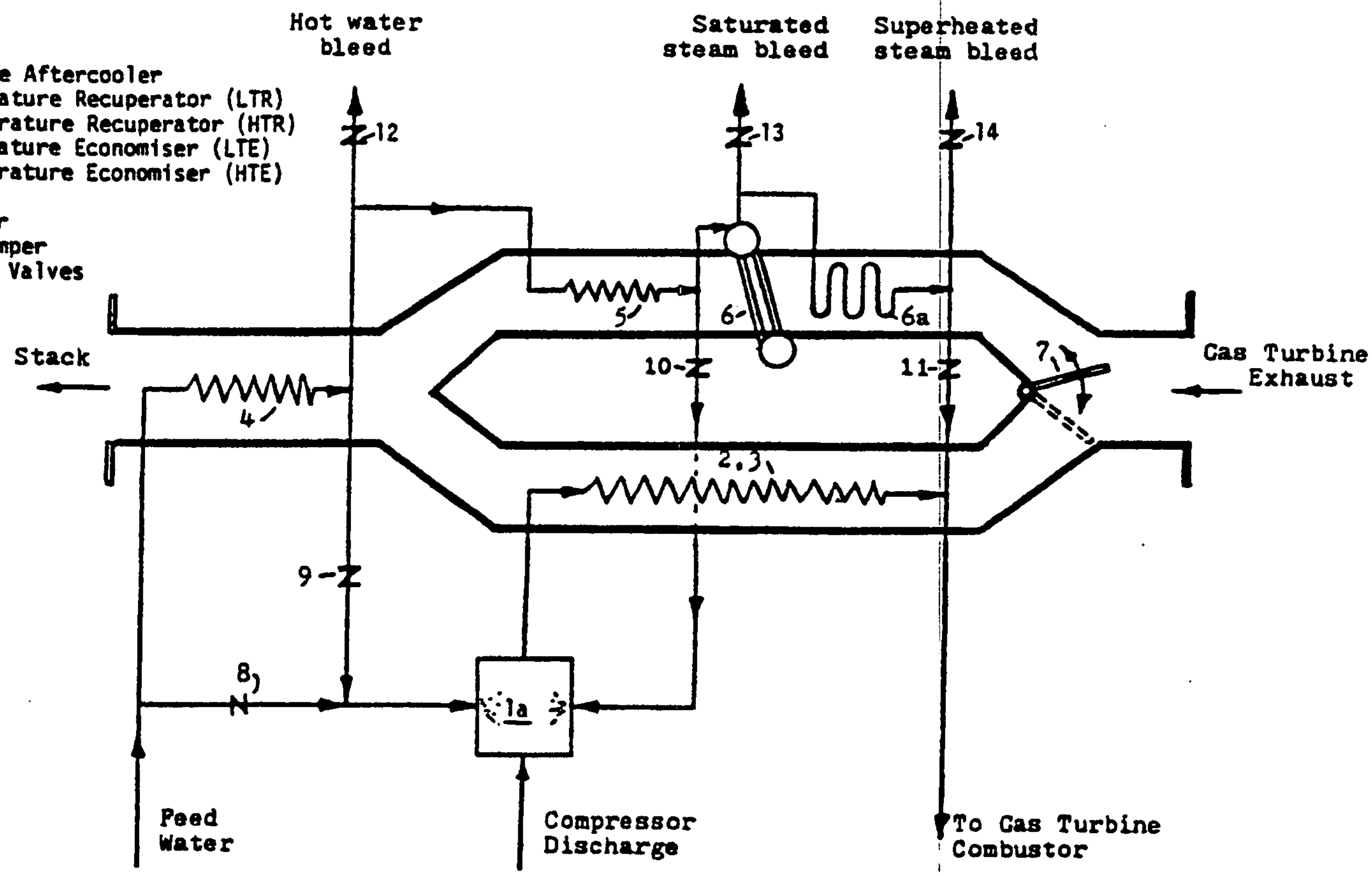

FIG. 9